STABILITY AND ROLL EFFECT OF THE STRAIGHT TRUCK SUSPENSION SYSTEM

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ABSTRACT: Safety and stability control especially those involving rollover motion are essential features of heavy vehicles. Therefore, this study concentrates on identifying the stability and roll effect on the front and rear straight truck suspension system. The straight truck roll motion is modeled in MATLAB/Simulink software and validated using TruckSim software. The behavior of the straight truck is identified when the truck right side tires hit the road bumps to generate the roll motion. The simulation results demonstrate that the rear suspension system is adequately more stable compared to the front suspension system based on the lower root mean square (RMS) value and shorter settling time.

KEYWORDS: *Roll Motion; Roll Effect; Stability Control; Straight Truck; Suspension System*

1.0 INTRODUCTION

The vehicle stability is measured based on the equilibrium of the dynamic motion response. The dynamic motions involved vertical, roll, pitch and yaw motions [1]. In most vehicle rides, vertical and roll motion are produced by the imbalance forces generated by the road disturbance. However, in a severe vehicle accident such as rollover condition is initiated from the roll motion. The rollover has a higher fatality rate rather than other types of vehicle collisions [2]. According to the Large Truck and Bus Crash Facts 2015, rollovers were reported in 15.6% of fatal truck accidents and 28.3% of injury crashes [3]. Furthermore, rollover is strongly associated with serious injuries to the truck driver. It was reported that 58% of fatal injuries to truck drivers is U.S. tractor-semitrailer accidents involved rollover crashes for 2011-2015 [4].

A rollover is a type of vehicle motion in which a vehicle tips over onto its side or roof due to rotation about the x-axis. A vehicle rollover is divided into two categories such as tripped and untripped [5]. The tripped rollover is caused by forces from an external object, such as a curb or a collision with another vehicle [6]. Untripped crashes are the result of steering input, speed and friction with the ground [7]. Untripped rollovers occur when cornering forces destabilize the vehicle. Before analyzing the rollover in depth, the effect on the stability and roll motion of the vehicle must be examined. This study aims to identify the stability and roll effect of the straight truck suspension system.

The suspension system which consists of springs, shock absorbers and linkages supports the frame of a vehicle [8]. Suspension system purposely contributing to the vehicle handling performance and braking for safety driving and remain the vehicle occupant comfortable from road vibrations and disturbances [9]. Instead of road disturbance, the imbalance of suspension parameter values contributes to the stability and roll effect of the vehicle. The front and rear suspension parameters of the straight truck are designed to be different values due to the load carried. In this study, the half straight truck is modeled in MATLAB/Simulink purposely to identify the behavior of the front and rear suspension in terms of front and rear body vertical and roll motions. This behavioral identification will be utilized in future study to improve the straight truck safety features mainly to prevent the rollover accident.

This paper is organized as follows: The first section contains the introduction and a review of some relevant preliminary works, followed by the modeling and simulation of the straight truck in the second section. The third section introduces the model verification using a TruckSim while the results of study in the fourth section. The last section contains the conclusion.

2.0 MODELING AND SIMULATION

This study starts with the derivation equation of motion of half straight truck model based on the schematic diagram of a half truck model with front and rear suspension system as shown in Figure 1 and Figure 2. These models consider that the two dimensional in vertical dynamics and two track half car model for roll dynamic [10].

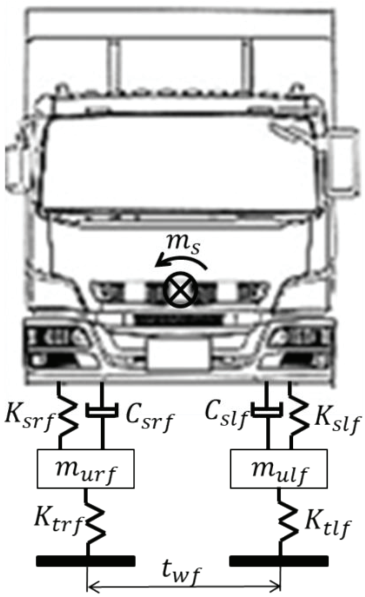

Figure 1: Schematic diagram of a half truck model with a front suspension system

Schematic diagram of a half truck model with a front suspension system as presented in Figure 1, contains the sprung mass, unsprung masses, springs, dampers, and tires. Thus, the equations of motion of a half truck model with front suspension system are derived based on the following equations:

$$m_s\ddot{z}_{sf} = -K_{slf}\left[\left(z_{sf} - \theta\left(t_{wf}/2\right)\right) - z_{ulf}\right] - C_{slf}\left[\left(\dot{z}_{sf} - \dot{\theta}\left(t_{wf}/2\right)\right) - \dot{z}_{ulf}\right] - K_{srf}\left[\left(z_{sf} + \theta\left(t_{wf}/2\right)\right) - z_{urf}\right] - C_{srf}\left[\left(\dot{z}_{sf} + \dot{\theta}\left(t_{wf}/2\right)\right) - \dot{z}_{urf}\right] \tag{1}$$

$$m_s\ddot{z}_{urf} = K_{srf}\left[\left(z_{sf} + \theta\left(t_{wf}/2\right)\right) - z_{urf}\right] + C_{srf}\left[\left(\dot{z}_{sf} + \dot{\theta}\left(t_{wf}/2\right)\right) - \dot{z}_{urf}\right] - K_{trf}\left(z_{urf} - z_{inrf}\right) \tag{2}$$

$$m_s\ddot{z}_{ulf} = K_{slf}\left[\left(z_{sf} - \theta\left(t_{wf}/2\right)\right) - z_{ulf}\right] + C_{slf}\left[\left(\dot{z}_{sf} + \dot{\theta}\left(t_{wf}/2\right)\right) - \dot{z}_{ulf}\right] - K_{tlf}\left(z_{ulf} - z_{inlf}\right) \tag{3}$$

$$I_s\ddot{\theta}_f = -K_{slf}\left[\left(z_{sf} - \theta\left(t_{wf}/2\right)\right) - z_{ulf}\right]\left(t_{wf}/2\right) - C_{slf}\left[\left(\dot{z}_{sf} - \dot{\theta}\left(t_{wf}/2\right)\right) - \dot{z}_{ulf}\right]\left(t_{wf}/2\right) + K_{srf}\left[\left(z_{sf} + \theta\left(t_{wf}/2\right)\right) - z_{urf}\right]\left(t_{wf}/2\right) - C_{srf}\left[\left(\dot{z}_{sf} + \dot{\theta}\left(t_{wf}/2\right)\right) - \dot{z}_{urf}\right]\left(t_{wf}/2\right) \tag{4}$$

where, m_s is the sprung mass; $\ddot{z}_{sf}$ is the front body vertical acceleration; z_{sf} is the front body displacement; $\dot{z}_{sf}$ is the front body vertical velocity; $\ddot{z}_{ulf}$ is the front left unsprung mass vertical acceleration; z_{ulf} is the front left unsprung mass displacement; $\dot{z}_{ulf}$ is the front left unsprung mass velocity; $\ddot{z}_{urf}$ is the front right unsprung mass vertical acceleration; z_{urf} is the front right unsprung mass displacement; $\dot{z}_{urf}$ is the front right unsprung mass velocity; K_{slf} is the front left spring stiffness; K_{srf} is the front right spring stiffness; C_{slf} is the front left damping coefficient; C_{srf} is the front right damping coefficient; I_s is the sprung mass moment of inertia; $\ddot{\theta}_f$ is the front body roll acceleration; K_{tlf} is the front left tire stiffness; K_{trf} is the front right tire stiffness and t_{wf} is the front track width.

Meanwhile, Figure 2 illustrates a schematic diagram of the half truck model with rear suspension system. In this study, the value of the parameter on the springs and dampers on the rear suspension are different from the front suspension system.

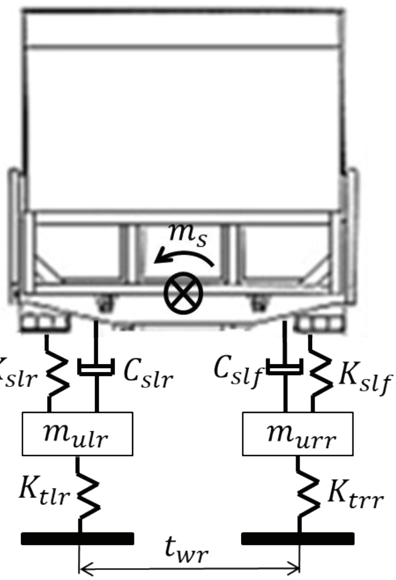

Figure 2: Schematic diagram of a half truck model with a rear suspension system

The following equations of motion of a half truck model with a rear suspension system are derived based on Figure 2.

$$\begin{aligned} m_s\ddot{z}_{sr} = -K_{slr}\left[\left(z_{sr} - \theta(t_{wr}/2)\right) - z_{ulr}\right] - \\ C_{slr}\left[\left(\dot{z}_{sr} - \dot{\theta}(t_{wr}/2)\right) - \dot{z}_{ulr}\right] - \\ K_{srr}\left[\left(z_{sr} + \theta(t_{wr}/2)\right) - z_{urr}\right] - \\ C_{srr}\left[\left(\dot{z}_{sr} + \dot{\theta}(t_{wr}/2)\right) - \dot{z}_{urr}\right] \end{aligned} \quad (5)$$

$$m_s\ddot{z}_{urr} = K_{srr}\left[\left(z_{sr} + \theta(t_{wr}/2)\right) - z_{urr}\right] + \tag{6}$$
$$C_{srr}\left[\left(\dot{z}_{sr} + \dot{\theta}(t_{wr}/2)\right) - \dot{z}_{urr}\right] -$$
$$K_{trr}(z_{urr} - z_{inrr})$$

$$m_s\ddot{z}_{ulr} = K_{slr}\left[\left(z_{sr} - \theta(t_{wr}/2)\right) - z_{ulr}\right] + \tag{7}$$
$$C_{slr}\left[\left(\dot{z}_{sr} + \dot{\theta}(t_{wr}/2)\right) - \dot{z}_{ulr}\right] -$$
$$K_{tlr}(z_{ulr} - z_{inlr})$$

$$I_s\ddot{\theta}_r = -K_{slr}\left[\left(z_{sr} - \theta(t_{wr}/2)\right) - z_{ulr}\right](t_{wr}/2) - \tag{8}$$
$$C_{slr}\left[\left(\dot{z}_{sr} - \dot{\theta}(t_{wr}/2)\right) - \dot{z}_{ulr}\right](t_{wr}/2) +$$
$$K_{srr}\left[\left(z_{sr} + \theta(t_{wr}/2)\right) - z_{urr}\right](t_{wr}/2) -$$
$$C_{srr}\left[\left(\dot{z}_{sr} + \dot{\theta}(t_{wr}/2)\right) - \dot{z}_{urr}\right](t_{wr}/2)$$

where, $\ddot{z}_{sr}$ is the rear body vertical acceleration; z_{sr} is the rear body displacement; $\dot{z}_{sr}$ is the rear body vertical velocity; $\ddot{z}_{ulr}$ is the rear left unsprung mass vertical acceleration; z_{ulr} is the rear left unsprung mass displacement; $\dot{z}_{ulr}$ is the rear left unsprung mass velocity; $\ddot{z}_{urr}$ is the rear right unsprung mass vertical acceleration; z_{urr} is the rear right unsprung mass displacement; $\dot{z}_{urr}$ is the rear right unsprung mass velocity; K_{slr} is the rear left spring stiffness; K_{srr} is the rear right spring stiffness; C_{slr} is the rear left damping coefficient; C_{srr} is the rear right damping coefficient; $\ddot{\theta}_r$ is the rear body roll acceleration; K_{tlr} is the rear left tire stiffness; K_{trr} is the rear right tire stiffness and t_{wf} is the front track width.

The half straight truck equations derived are modeled in MATLAB/Simulink software as shown in Figure 3. The equations derived involved sprung mass, unsprung masses, tires, springs and dampers. The input data for the ride test is 10 cm step input with a speed of 90 km/h. The model of half straight truck is developed in MATLAB/Simulink software starts with the parameter identification. The parameters used in this simulation are set based on Chaganti et al. [11]. All parameters are given in Table 1. Heun solver has been applied to simulate the tractor semi-trailer combinations model [12].

The outputs of the simulation are the body vertical acceleration and body roll acceleration.

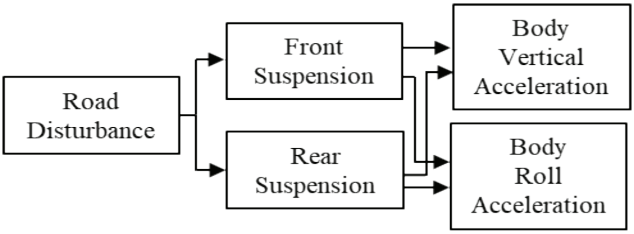

Figure 3: Half straight truck model in MATLAB/Simulink

Table 1: The straight truck parameters

Parameters	Values
m_s (kg)	12487
m_{ul} (kg)	1706
m_{ur} (kg)	1706
I_s (kg.m^2)	24201
$K_{sl,r,front}$ (N/m)	378666
$K_{sl,r,rear}$ (N/m)	1083004
$C_{sl,r,front}$ (Ns/m)	260000
$C_{sl,r,rear}$ (Ns/m)	200000
K_{tl} (N/m)	1000000
K_{tr} (N/m)	1000000
$t_{w,front}$ (m)	2.04
$t_{w,rear}$ (m)	1.85

3.0 MODEL VERIFICATION

In this section, the half truck model is verified using the TruckSim software [13]. The TruckSim is a multi-body dynamics software used in the heavy truck industry to simulate the vehicle dynamic characteristics of an actual truck and verify the model developed using a mathematical derivation [14]. Figure 4 shows the vehicles dataset in TruckSim which contains the standard examples of various truck and combination vehicle configurations.

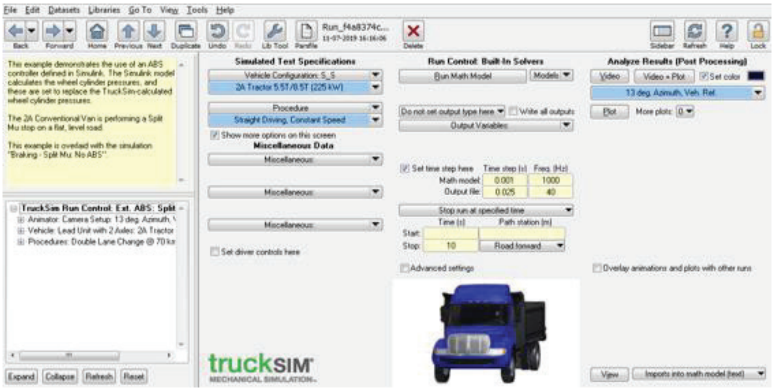

Figure 4: TruckSim vehicle dataset

The verification of the straight truck model developed in this study is conducted using a two axle's dump truck with laden condition. The ride test procedure is performed with constant speed of 90 km/h. In this procedure, the truck hit the 10 cm bump on the right side to generate the roll motion on the truck. The illustration of the ride test is presented in Figure 5.

Figure 5: Straight truck ride test in TruckSim

4.0 RESULTS AND DISCUSSION

This section discusses the behavior of the straight truck model responses when the truck right side tires hit the 10 cm road bumps. Figure 6 presents the body vertical acceleration responses of the half straight truck model affected by the configuration of front and rear suspension system respectively. It is observed that maximum body vertical acceleration generated at the rear body is 3.04 m/s^2 while front body vertical acceleration is 1.862 m/s^2. The lower front body vertical acceleration compared to rear body leads to increase the ride comfort to the driver [15]. However, the settling time response generated at rear body is 8 seconds compared to the front body at 15 seconds. The maximum vertical acceleration with shorter settling time is generated at rear body because of the higher value of spring stiffness is utilized at the rear suspension [16]. This is due to the fact that the rear suspension is supported a total load of goods brought by the truck.

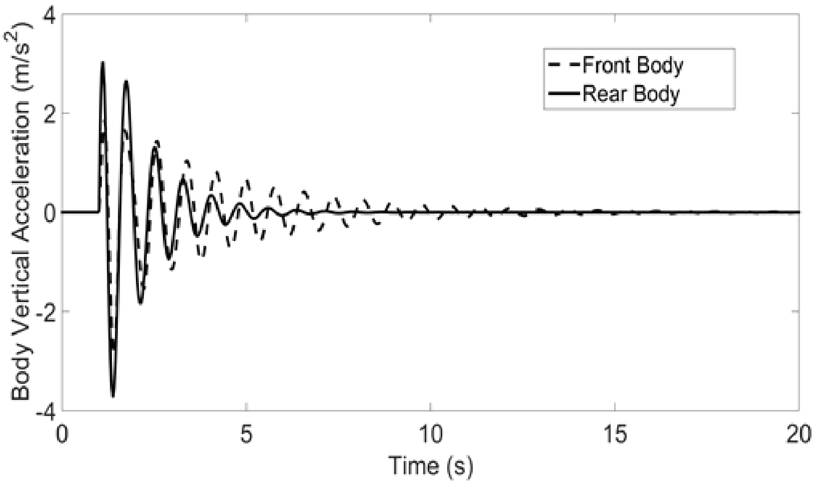

Figure 6: Straight truck body vertical accelerations

A similar response is observed for body roll acceleration as depicted in Figure 7. The maximum roll acceleration generated at rear body is 0.88 rad/s^2 while front body roll acceleration generated 0.46 rad/s^2. The highest roll acceleration responses were generated at rear body. It is due to the high value of spring stiffness at the rear suspension system. Simultaneously, the settling time for front and rear body roll acceleration is similar to the vertical acceleration. Henceforth, it is observed that the settling time between body vertical and roll acceleration is synchronized in order to achieve optimum stability.

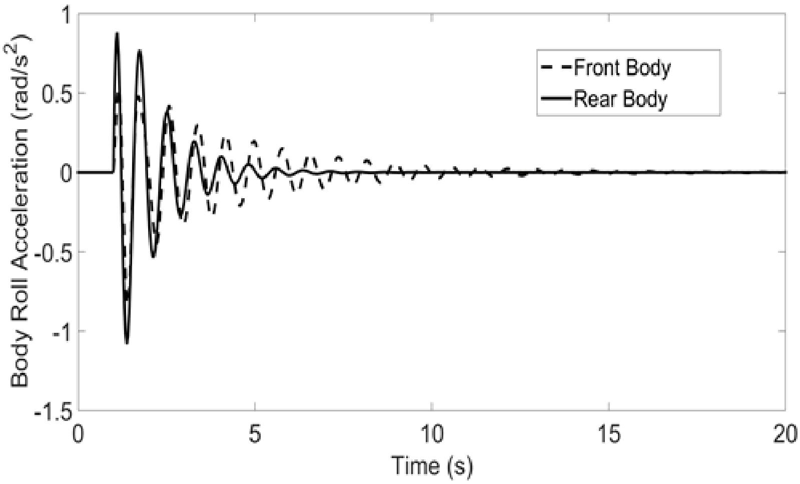

Figure 7: Straight truck body roll accelerations

Table 2 presents the root mean square (RMS) value between the front and rear body of vertical and roll acceleration. The RMS values of these two responses were observed to quantify the ride performance of the straight truck [17]. From Table 2, it is observed that the RMS value for front body vertical acceleration is 4.695×10^{-3} m/s^2 while rear body vertical acceleration is 3.957×10^{-4} m/s^2. Furthermore, the RMS value for front body roll acceleration is 1.360×10^{-3} rad/s^2 while rear body vertical acceleration is 1.146×10^{-4} rad/s^2. The lower RMS value indicates that the ride performance of rear body is better than the front body. This also supported by Lai and Liao [18] that the better system performance is produced from the lowest RMS value.

Table 2: RMS values of simulation results on body vertical and roll acceleration response

Performance Criteria	Front	Rear
Vertical Acceleration (m/s^2)	4.695×10^{-3}	3.957×10^{-4}
Roll Acceleration (rad/s^2)	1.360×10^{-3}	1.146×10^{-4}

5.0 CONCLUSION

In conclusion, the stability and roll effect of the straight truck suspension system were identified. It shows that rear suspension system performance is better than the front suspension system due to the lower RMS value and shorter settling time. Thus, this condition provides a comfortable and safe driving to the driver.

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